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Simulation studies of refrigeration cycles: A review

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A B S T R A C T

Presently out of the total energy consumption, a large share of energy is being used by refrigeration and air conditioning equipments. The present study is based on literature review on the refrigeration systems, currently used refrigerant–absorbent pairs and also on different sources of energy. The basis of this study is to know about the user friendly softwares used for the simulation techniques and also on the scope of different alternative forms of energy as a source to generator. The effects of operating temperature, effectiveness of heat exchangers and choice of working fluid on the systems were studied. It is evident from the studies that the cycle performance (COP) improves with increasing generator and evaporator temperatures, but reduces with increasing the absorber and condenser temperatures. The use of heat exchangers improves the overall performance of the system, especially solution heat exchanger (SHE). It is also evident that solar energy obtained in the range of about 100 °C is having good potential to supply sufficient energy to the generator for absorption–refrigeration cycles.

Contents

1. Introduction

The depletion of the premium energy sources however has a universal, overall impact and it adversely affects all commercial as well as our daily lives. The increase in the energy demand leads to the increase in its supply; however, the increase energy demand poses a pressure on the energy sources which are mostly conventional and

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Nomenclature

FR	Flow ratio
PTC	Parabolic trough collector
SCP	Specific cooling power (Wkg^{-1})
SHE	Solution heat exchanger
RHE	Refrigerant heat exchanger
SRHE	Solution-refrigerant heat exchanger
DT	Temperature difference in the cascade heat exchanger (K)
X	Exergy lost rate (kW)
G	Generator
EDR	Exergy destruction rate
E	Evaporator
C	Condenser
A	Absorber
X	Solution concentration
T	Temperature
Q	Heat transfer rate (kW)
ϵ	Effectiveness of heat exchanger
Gc	Gas cooler
η	efficiency
P	Pressure
UA	Overall heat transfer coefficient—area product (WK^{-1})
CHE	Cascade heat exchanger

Subscripts

CO_2	Carbon-dioxide
NH_3	Ammonia
Comp	Compressor
Cond.	Condenser
Evap., ev	Evaporator
St	Storage
Heat	Heating
1, 2, 3, etc.	State points
u	Internal energy ($kJkg^{-1}$)
i	Inside
o	Outside

hence their exploration increases which would gradually deplete the energy resources pool if immediate saving steps are not taken. Also with the increase in energy demand, the prices also start off showing many folds toll and this has raised an alarm to save energy, by using minimum energy from the conventional resources like crude-oil, natural gas, oil shale etc. The ever increasing energy demand can be reduced to some extent by using energy sources other than conventional sources like solar energy, biomass, geothermal energy, wind energy etc. This ever increasing pressure of the energy consumption is more prominent in the industrial sector and is mostly related to the heavy inductive machinery like motors, refrigeration and air-conditioning units etc. These refrigeration and air-conditioning units contribute to the major share of the energy consumption because of heavier compressor used in such systems. This problem now is being tried to be removed by the usage of the heat or thermal energy driven absorption systems. Refrigeration and air conditioning (RAC) play a very important role in modern human life for cooling and heating requirements. This area covers a wide range of applications starting from food preservation to improving the thermal and hence living standards of people. The utilization of these equipments in homes, buildings, vehicles and industries provides for thermal comfort in living/working environment and hence plays a very important role in increased industrial production of any country. Due to the increasing demand of energy primarily for RAC and HP applications (around 26–30%), there is degradation of environment, global warming and depletion of ozone layer, etc. Therefore to overcome these aspects there is urgent need of efficient energy utilization methods apart from waste heat recovery for useful applications.

As energy conservation is becoming an increasingly important aspect/parameter, there is a need to optimize the thermodynamic processes for the minimum consumption of energy. Many parameters affect the performance of a refrigeration cycle and in order to optimize their design, a thorough study based on the second law of thermodynamics (exergy analysis) analysis is required. Although, first law of thermodynamic analysis method is most commonly used, however, this is concerned only with the conservation of energy and therefore it cannot show how or where irreversibility in the system and or a process occurs. On the other hand, second law based exergy analysis is another well known method being used to analyze these cycles. Unlike, the first law, second law analysis determines the magnitude of irreversible processes in a system and thereby, provides an indication to point out the directions in which the engineers should concentrate more to improve the performance of the thermal system.

The proper management of the energy in any industrial process involves the reduction or removal of the conventional energy used and economy can be improved by the reuse of the waste heat coming from the processing steps. By wisely using such recoverable heat from the processor and to switching over to some readily available energy sources is a better way to cut the stress on the conventional energy sources.

Continuous efforts have been made by numerous researchers on different types of heat pumps in order to improve their performance and also to make them cost effective [1–11]. Building, industry and transportation are three main commercial energy consumers. The building accounts for 30% of the total energy use in some industrialized countries. Saving energy and reducing green house gas emissions are given more and more attention and thus decreasing the energy consumption of the HVAC systems, responsible for more than half of the building energy use is imperative. The energy consumption in buildings, commercial installations and space air conditioning constitutes a huge share of the total energy consumption not only in the developed but also in the developing countries. Facing the ever increasing pressure of energy demand, environmental degradation, global warming and the depletion of the ozone layer due to various reasons, most commonly industrialization, the efficient use of the energy is a hot topic of research after the Kyoto and Montreal Protocol. This, in turn, can help in the long run for solving the huge environmental degradation and global warming problems. Under pressure from the scientific community some governments have decided to restrict the wastage in different forms by means of penalty on the industries for the release of high temperature waste hot water and residual waste heat in the open environment. A large number of people are concerned about the heat recovery from the waste hot water before it is released to open environment that may be harmful to different types of plants, animals, species, and hence, may cause the ecological disturbance. In such a case, the heat pump system can be used to extract the heat from waste hot water to utilize it for other industrial applications, such as in dye industry, heating and cleaning and so on. This not only can utilize the waste heat for higher temperature applications but also can reduce a huge potential of the environment degradation. It is well known that there is a huge

potential of energy/heat recovery from the industrial waste heat. This will not only be helpful in saving energy resources but also can decrease the ozone depletion and release of greenhouse gases.

The present communication describes detailed mathematical models, simulation computer program and their application for cooling as well as heating purposes. The models have been developed utilizing distributed methods which are necessary to predict accurately the great variations of both refrigerant's thermo-physical properties and local heat transfer coefficients during the processes. They also enable performance analysis for different arrangements and the influence of some important operating and design parameters. These systems are feasible to run on the heat obtained from the waste hot water from various industries as well as from the energy obtained from non-conventional energy resources like solar energy, biomass, etc.

2. Analysis of different types of refrigeration cycles

2.1. Thermodynamic evaluation of a modified aqua-ammonia absorption refrigeration system

Heat released during absorption process is utilized to heat the strong solution coming out of absorber; therefore the heat required for further heating the solution in the generator is considerably reduced and hence yields enhanced coefficient of performance.

Kumar and Kaushik [12] carried out an analysis of modified aqua-ammonia absorption system to study the effects of various operating parameters and effectiveness of heat exchangers on COP and various heat/mass transfer states. They found that improvement of about 10% in the COP for absorption cycle is achieved at an evaporator temperature of 0 °C and large amount of heat is evolved in the absorber during the absorption process. Thermodynamic analysis based on mass, material and heat balance for a modified aqua-ammonia absorption–refrigeration system was carried out. A detailed parametric study of modified absorption cycle has been undertaken to obtain optimum operating condition during the system operation. The model is considered as a combination of the components: absorber, heat exchanger and the externally cooled absorber. In absorber heat exchanger, the weak solution which is cooled to the solution condition in the heat exchanger is spread to absorb the ammonia vapor coming from the evaporator. The strong solution coming out of the absorber heat exchanger gets preheated with the help of heat of mixing in the absorption side of the absorber heat exchanger. The requisite temperature difference, between the hot and cold stream in absorber heat exchanger, is maintained by external absorber cooling. Preheated strong solution gets further heated in the solution heat exchanger where vapor generation also takes place to some extent. This reduces the heat input to generator for complete vapor generation process and hence, the COP of the cycle increases. The thermodynamic analysis of the modified aqua-ammonia absorption–refrigeration cycle is based on mass, material and heat balance along with the following assumptions:

- Pressure in the generator and refrigerator is equal to the condenser pressure.
- Low pressure level in the system is equal to the pressure in the absorber and the strong solution leaving the absorber is heated to saturation condition.
- Mass flow rate and concentration of the refrigerant in the vapor leaving the rectifier is assumed to be constant.

Computer modeling of the modified aqua-ammonia absorption system is based on thermodynamic state equation and has been carried out to study the effect of various design parameters on the COP. Variation of COP with the generator temperature is studied and two values of 0 °C and −10 °C for the evaporator temperature are chosen. The results revealed that the variation of COP with the generator temperature is higher at higher condenser temperature.

2.2. Thermodynamic studies on HFC134a-DMA double effect and cascaded refrigeration systems

The double-effect VARS consists of components similar as in single stage absorption system with an additional generator. Ideally, this system has three pressure levels and five operating temperature levels which are T_e, T_a, T_c, $T_{g\text{-I}}$ and $T_{g\text{-II}}$. Low pressure prevails in the evaporator and the absorber, high pressure in the generator-I and intermediate pressure in the condenser and generator-II. The cascade system consists of two single-stage VARS which are coupled so that the evaporator of the first stage works as condenser of the second stage. Due to the lower condensing temperature at the second stage, lower evaporating temperatures can be achieved compared to the single-stage VARS. The analysis of model is carried out by Songara et al. [13] and is based on the following assumptions for the thermodynamic analysis, which are common to both systems:

- The absorbent is non-volatile in the temperature range considered.
- Pressure changes occur only at pump and pressure-reducing valves.
- The weak solution leaving the generators and strong solution leaving the absorbers are at equilibrium conditions.
- The condition of refrigerant at the exit of evaporator and condenser is saturated.
- The solution pump work is neglected since the thermodynamic work required to pump a liquid is very small due to negligible change in specific volume of liquid.
- The effectiveness of solution heat exchangers is 0.85 which is a practically feasible value.

The heat quantities are calculated for 1 kW of refrigeration capacity at the evaporator. The heat input at generator, circulation ratio, second-law efficiency and COP are the main performance parameters and are calculated using the mass and energy balance equations and the thermodynamic properties of HCFC134a-DMA were taken from the work of Borde and Jelinek [14]. The operating temperature ranges considered are given in Table 1. Variations in operating parameters are calculated in order to analyze the performance of these conditions on the system and final analysis has been drawn in the way that Double-effect HFC134a-DMA absorption systems yield COP comparable to single-stage HCFC22-DMA systems, but at slightly higher heat source temperatures. In order to obtain refrigeration at sub-zero temperatures at low heat source temperatures, one may have to resort to the cascade systems.

Table 1
Operating temperature values.

Double-effect system	Cascade system
Evaporating temperature Te = 0–10 °C	Evaporator e_2 temperature = −10-0 °C
First generator temperature T_{gI} = 100–125 °C	Evaporator e_1 temperature = 0–20 °C
Second generator temperature $T_{g||}$ = 60–75 °C	Condenser c_1 temperature = 15–40 °C
Heat rejection temperatures T_c and T_a are held constant at 25 °C.	Absorber a_1 and Absorber a_2 temperature = 15–40 °C Generator g_{I} and g_{II} temperature = 90 °C

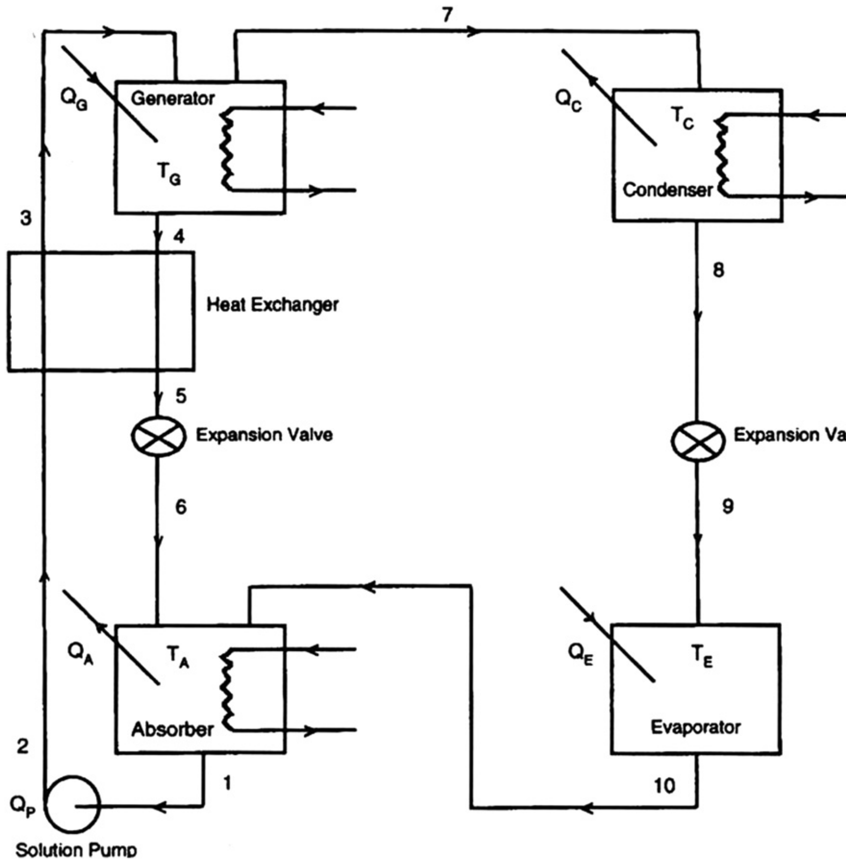

Fig. 1. A schematic diagram of ACS. Dincer and Dost, [15].

2.3. A simple model of absorption cooling systems (ACSs)

Absorption systems are usually operated by steam or hot water generated from natural gas, oil-fired boilers and electrical heaters. The direct burning of coal and natural gas as an energy source hampered the application of ACSs, but at the same time opened up the other opportunities like usage of heat delivered from the solar collectors to operate the ACS. With the increase in energy costs, the low temperature heat (90–100 °C) earlier rejected from the process plants to the atmosphere can now easily be used to operate the absorption cooling systems, providing a cooling effect. LiBr–H_2O and NH_3–H_2O have been well known working fluids for a long time and NH_3–H_2O ACSs are used for all refrigeration purposes.

Dincer and Sadik [15] introduced a simple expression for analyzing mass and heat transfer in LiBr–H_2O ACSs. The schematic diagram of the system is shown in Fig. 1. A majority of the experimental and theoretical analysis on the ACSs have been related to the LiBr–H_2O and NH_3–H_2O working fluids. Some of these studies have been undertaken by many researchers [16–22] and for this the knowledge of the properties of the working fluids and the knowledge of the major characteristic parameters like the performance, cost and size of the refrigeration unit is very important. However, a number of studies have been carried out by many researchers [23–31] on the absorption refrigerant–absorbent combinations and the results of these working fluids have been presented as R123a-ETFE, R22-DMETEG, R123a-DTG, and R123a-DTrG. The system performs under following assumptions:

- The ACSs are single stage system,
- steady state conditions exists,
- refrigerant (water) leaves the generator in pure vapor-phase,
- pressure drops in the components are negligible.

For the detailed analysis of the ACSs a great help has been taken from many authors [27,28,29]. The simplicity in the equations for various state points/components can be obtained by considering the optimum design parameters given by the authors [28,31], such as T_E=4.5 °C, T_C=T_A=30 °C, ϵ=0.9 and for generator temperature T_G=90 °C and 100 °C.

For comparison purposes both the temperature values for generator are considered and the concentration values for LiBr on the mass basis were taken as X_a=0.685 and 0.695 and X_s=0.5 to avoid crystallization. Using these optimum conditions, different temperature values are computed using a computer program partly based on ASHRAE [32] standards. Upon using the values of the temperature, enthalpies and pressure in ASHRAE [32], the equations in terms of the mass flow rates of the weak solution corresponding to the two generator temperature 90 °C and 100 °C are obtained and the results are plotted graphically as shown above in Figs. 2 and 3, respectively.

The results show that COP of LiBr–H_2O ACSs operating at the generator temperature of 90 °C becomes more efficient in providing the evaporator temperature of 4.5 °C and this is because of the lower generator temperature which provides the same evaporator temperature and thus this shows that the solar energy is an appropriate source of energy for the above considered system.

2.4. Theoretical analysis of vapor compression refrigeration system with R502, R404A, R507A

Arora and Kaushik [33] carried out the detailed exergy analysis of an actual vapor compression refrigeration (VCR) cycle. A computational model has been developed for computing coefficient of performance (COP), exergy destruction, exergetic efficiency and efficiency defects for R502, R404A and R507A. The investigation

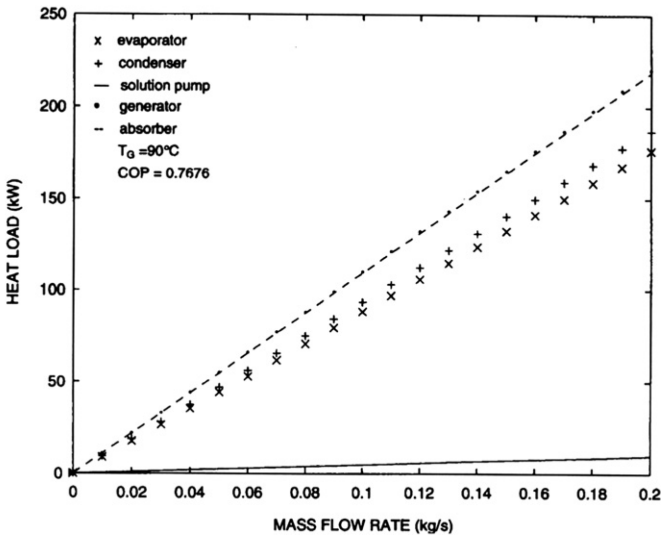

Fig. 2. Variations of the absorber, pump, generator, condenser and evaporator loads against mass flow rates of weak solutlon at T_G=90 °C. Dincer and Dost [15].

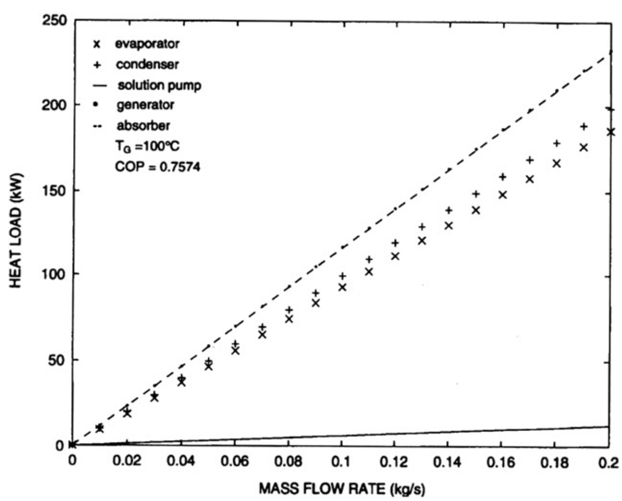

Fig. 3. Variations of the absorber, pump, generator, condenser and evaporator loads against mass flow rates of weak solutlon at T_G=100 °C. Dincer and Dost [15].

has been done for evaporator and condenser temperatures in the range of −50 °C to 0 °C and 40–55 °C, respectively. Various researchers, like Aprea and Mastrullo [34], Camporese et al. [35], Gunther and Steimle [36], Sami and Desjardins [37], have suggested different HCFC, HFC and HC blends as potential substitutes for R502 and compared the performance of these substitutes either theoretically or experimentally. In some of these studies, measurements of thermodynamic and thermo-physical properties were carried out. In one of the recent studies, Xuan and Chen [38] experimentally tested HFC-161 mixture (HFC-161, HFC-125 and HFC-143a (10/45/45 wt%)) as an alternative refrigerant to R502. Their results substantiated that this new refrigerant can achieve a high level of COP than R404A and R507 and can be considered as a promising retrofit refrigerant to R502. Fig. 4 represents the system diagram.

A computational model is developed for carrying out the energy and exergy analysis of the system using Engineering Equation Solver software [39]. The measure of performance of refrigeration cycle is coefficient of performance (COP) which is based on the first law of thermodynamics. The second law of thermodynamics derives the concept of exergy, which is the measure of usefulness, quality or

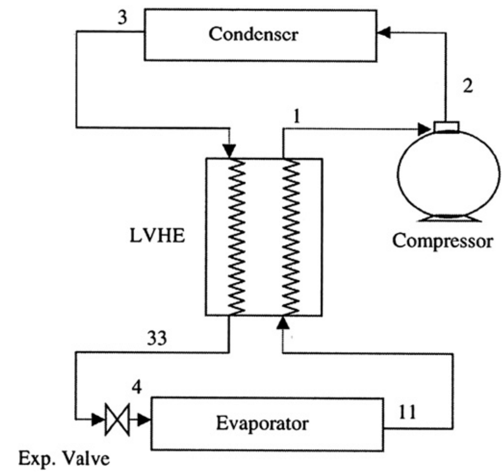

Fig. 4. A Schematic diagram of a vapor compression cycle with liquid vapor heat exchanger. Arora and Kaushik [33].

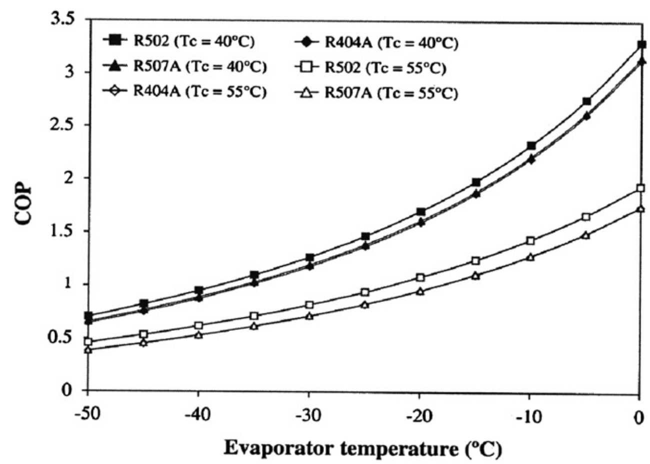

Fig. 5. Variation of COP with evaporator temperature. Arora and Kaushik [33].

potential of a stream to cause change and an effective measure of the potential of a substance to impact the environment [40], which always decreases due to thermodynamic irreversibility. The input data assumed for the computation of results shown in Fig. 4 is given below:

- Mass flow rate of refrigerant=1 Kg s^{-1}
- Degree of sub cooling of refrigerant=5 °C
- Isentropic efficiency of the compressor=0.75
- Difference between evaporator and space temperarture=15 °C
- Pressure drop in evaporatior=20 Kpa
- Pressure drop in a condenser=10 Kpa
- Dead state temperature=25 °C

Fig. 5 shows the comparison of coefficient of performance against evaporator temperature. The increase in evaporator temperature decreases the pressure ratio across the compressor which in turn reduces compressor work and eventually the cooling capacity increases because of increase in specific refrigerating effect. The combined effect of these two factors is to enhance the overall COP.

Fig. 6 represents the effect of evaporator temperature on EDR and exergetic efficiency. The rise and fall of the exergetic efficiency, with increase in evaporator temperature, are based on two parameters. First parameter is exergy of cooling effect, i.e. Q_e, with increase in evaporator temperature, Q_e increases and

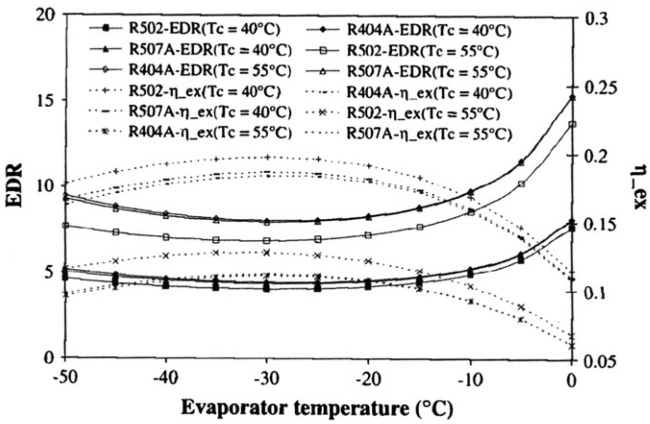

Fig. 6. Variation of EDR and exergetic efficiency with evaporator Temperature. Arora and Kaushik [33].

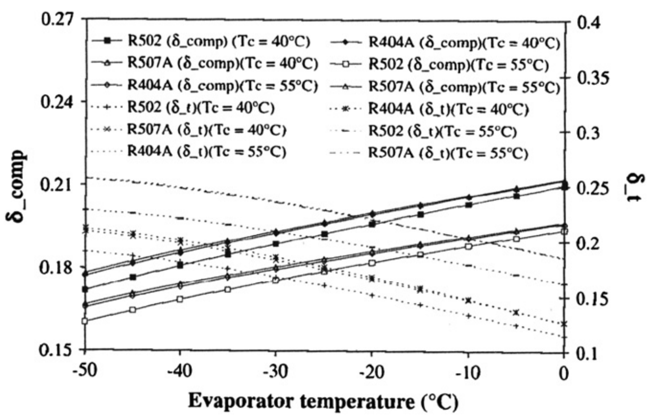

Fig. 7. Comparison of efficiency defects in compressor (δ_comp) and throttle value (δ_t) with evaporator temperature. Arora and Kaushik [33].

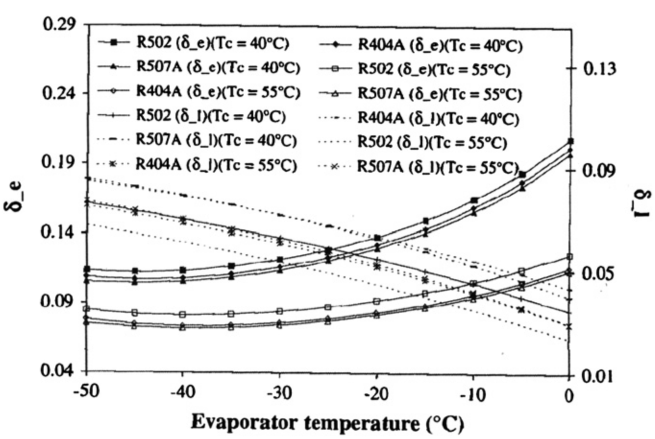

Fig. 8. Efficiency defect in evaporator (δ_e) and liquid vapor heat exchanger (δ_1) with evaporator temperature. Arora and Kaushik [33].

the second being the compressor work which decreases with increase in evaporator temperature. Both Q_e and W_c have positive effect on increase of exergetic efficiency.

The results of efficiency defects in the compressor, condenser, throttle valve and evaporator are depicted in Figs. 7 and 8. The efficiency defect gives the idea of fraction of input energy destroyed in a particular component and also assists in pointing out the worst component of a system. It is apparent that the most efficient component of the vapor compression system is liquid vapor heat exchanger in which minimum efficiency defect is observed (between 2% and 8.5%). In the descending order of efficiency defects, the components can be arranged in the sequence as—condenser > compressor > throttle valve > evaporator > liquid vapor heat exchanger. It is necessary to cite here that the input energy consumed due to irreversibility in condenser varies between 25% and 43%.

Arora and Kaushik [33] carried out an extensive energy and exergy analysis of R502, R404A and R507A in an actual vapor compression cycle and concluded that

1. COP and exergetic efficiency for R507A are better than that for R404A at condenser temperatures between 40 °C and 55 °C. However, both refrigerants show 4–17% lower value of COP and exergetic efficiency in comparison to R502 for the condenser temperatures between 40 °C and 55 °C. In the descending order of EDR, these refrigerants can be arranged as R404A, R507A and R502.
2. The worst component from the viewpoint of irreversibility or exergy destruction is condenser followed by compressor, throttle valve and evaporator.
3. The increase in dead state temperature has a positive effect on exergetic efficiency and EDR, i.e. EDR reduces and exergetic efficiency goes up with increase in dead state temperature.

2.5. Performance studies on mechanical adsorption hybrid compression refrigeration cycles with HFC 134a (double effect)

Banker et al. [41] carried out an investigation on the efficacy of hybrid compression process for refrigerant HFC-R134a in cooling applications. The conventional mechanical compression is supplemented by thermal compression using a string of adsorption compressors. Activated carbon is the adsorbent for thermal compression segment. The alternatives of bottoming either mechanical or thermal compression stages are also investigated. They showed that almost 40% energy saving is realizable by carrying out a part of the compression in a thermal compressor compared to the case when the entire compression is carried out in a single-stage mechanical compressor. The hybrid compression is feasible even when low grade heat is available. The vapor compression refrigeration (VCR) with the positive displacement compressors (such as reciprocating, rotary, and scroll or screw compressors) continues to be the choice of cooling demands. The most investigated combination is the lithium bromide and water system, which cannot be used below about 5 °C, and is handicapped by operation at sub-atmospheric pressures. The other most widely investigated pair, ammonia and water, has the problem of carryover of water vapor into the refrigeration circuit, need for high pressures and reservations on acceptability of ammonia in small scale refrigeration units. New developments in scroll and screw compressor technologies have considerably reduced the mechanical energy requirements. Adequate advances have been made in system practices using this refrigerant. A cascade refrigeration system which reduced the energy consumption of a HCFC-22 compression system with ammonia–water absorption system provided a clue to possible energy saving [42]. When refrigeration is a part of a process industry, wherein considerable amount of waste heat is available or where other low grade energy sources, such as solar energy are also available, such a combination of compression processes could be even more advantageous.

The schematic diagram is shown below in Fig. 9. The performance indicator is calculated in terms of conventional COP and will be retained as the first indicator. The data required for the analysis are the equation of state for HFC 134a (Tillner-Roth and Baehr [43], adsorption characteristics of activated carbon and HFC134a system [44]). The entire calculation scheme was

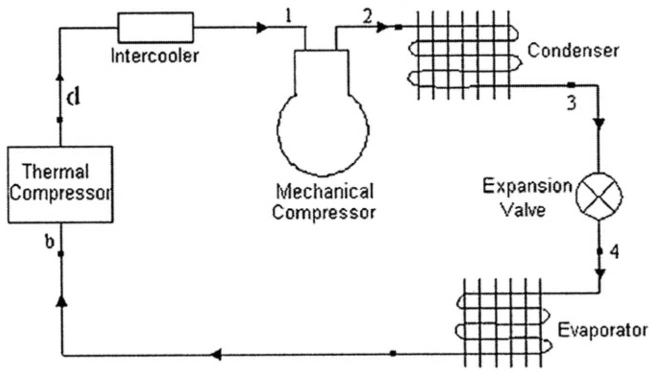

Fig. 9. A schematic diagram of adsorption+compressor hybrid refrigeration system. Banker et al. [41].

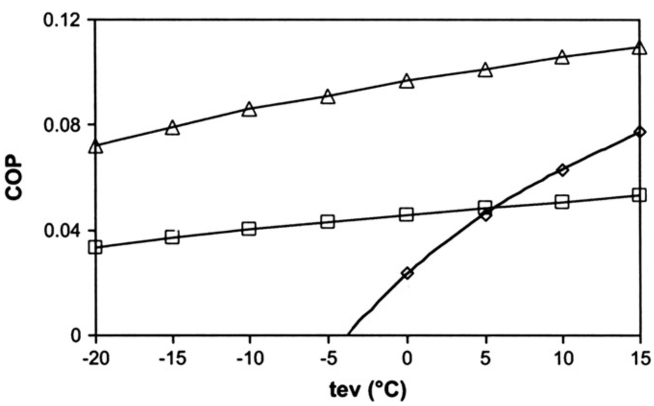

Fig. 10. Coefficient of performance at various evaporator temperatures single stage (◇), two-stage (□) and hybrid cycles (△). Banker et al. [41].

programmed on a MATLAB platform. A comparison is made between single and two stage thermal compression and hybrid compression processes. The calculation scheme covered a number of condensing/adsorption, evaporating and desorption temperatures.

Fig. 10 shows comparison of COP for single stage, two-stage thermal compression and hybrid systems. It is depicted from the results that COP of a two-stage system almost remains constantly fairly uniform over the entire range of evaporating temperatures investigated; single-stage system is better than the two-stage one for $t_e > 5$ °C and COP of the hybrid cycle is the highest because of less energy needed for the second stage which is mechanical compression. When the exergy components of useful cooling and the heat supplied to the thermal compressor are considered, a better appreciation of the thermal processes can be obtained.

Fig. 11 shows a comparison of uptake efficiencies for single-stage, two-stage, hybrid and vapor compression cycle (volumetric efficiency). The case of packing effectiveness for single stage thermal compression is shown to indicate the qualitative improvements. For the first as well as the second compressors in a two stage system, and the hybrid system the uptake efficiencies are approximately equal to the variation of volumetric efficiency which is similar to uptake efficiency and is also plotted against evaporating temperatures. It can be seen that the hybrid compression improves the uptake efficiency of the thermal stage significantly. Indeed, a single stage thermal compression would not have been possible for $t_e < -8$ °C and there would be only a marginal reduction even if the packing density is increased from 325 to 450 kg/m^3. The comparison of shaft power requirement for mechanical stage of hybrid cycle single-stage mechanical compressor is made in Fig. 12.

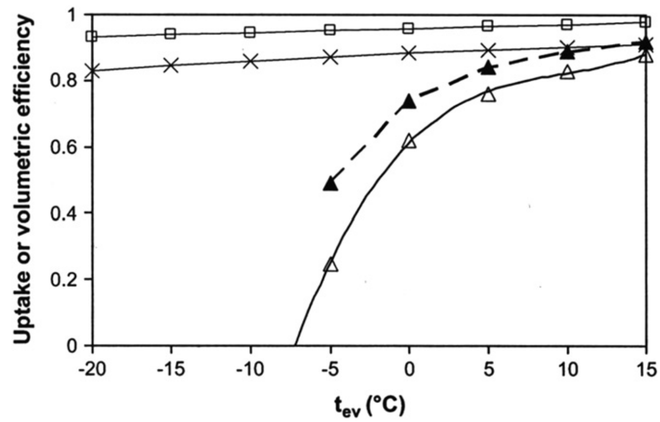

Fig. 11. Uptake efficiency variation for single stage packing efficiency 325 Kg/m^3 (△) and 450 Kg/m^3 (▲), two stage and thermal stage of hybrid cycle (□) for packing efficiency=325 Kg/m^3 and mechanical compressor in VCR(x). Banker et al. [41].

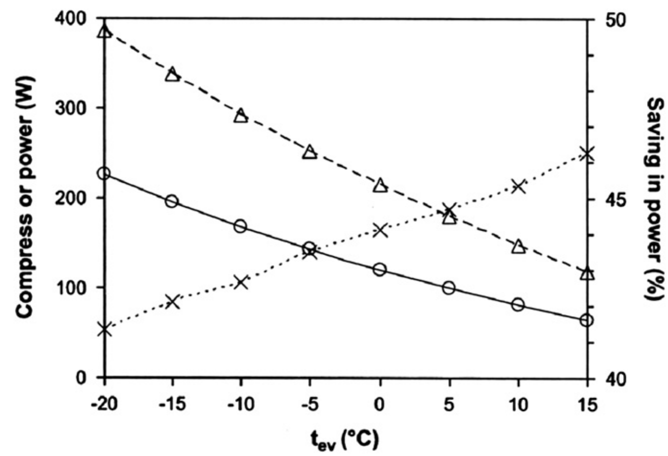

Fig. 12. Assessment of compressor power in hybrid and single stage mechanical compression cycles (△), single stage mechanical; ○, hybrid mechanical; *x*,% saving [right ordinate]. Banker et al. [41].

It is seen that a saving of power $>40\%$ is possible. However, the absolute saving decreases marginally with reducing evaporating temperatures. Thus, hybrid compression can be a potential avenue of energy conservation in the refrigeration systems.

2.6. Theoretical analysis of CO_2 ammonia cascade refrigeration system for cooling application at low temperature

Amongst the natural refrigerants, Lorentzen and Petterson [45] suggested the use of CO_2 and it seems to be a promising natural refrigerant [46–50]. The key advantages of CO_2 are that it is non-toxic, non-explosive, easily available, environmentally friendly and has excellent thermo-physical properties. The use of the CO_2 in the low temperature stage and ammonia in the high temperature stage turn out to be an excellent alternative for low temperature cooling [51,52]. Kim et al. [53] summed up the improvements in the performance of the system based on CO_2 and their applications. They provided a critical review of literature and discussed important trends and characteristics in the development of the CO_2 technology in refrigeration system. Alberto et al. [54] did the analysis and scope of their research work is focused on the evaluation of design parameters on the operation of CO_2/NH_3 cooling system and its influence over the COP of the system and exergetic efficiency. The system

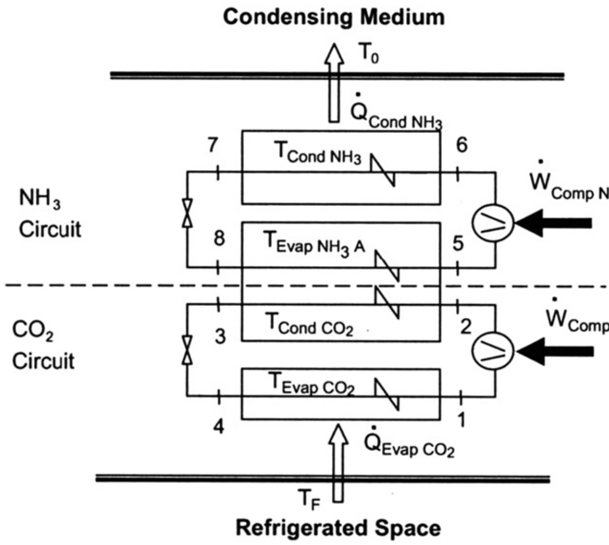

Fig. 13. Schematic diagram of CO_2/NH_3 cascade refrigeration system. Dopazo et al. [54].

description is shown in Fig. 13. It consists of the two single stage system connected by a heat exchanger. The low temperature system uses CO_2 as the refrigerant for cooling and high temperature system uses ammonia as the refrigerant. The analysis is based on the mathematical model to calculate the compressor's power, heat transfer rates and exergetic and energetic efficiencies. Considering each component of the cascade system for a stationary flow mass balance, energy balance and exergy balance have been performed. They made the following assumptions for their analysis:

- Refrigerant at the cascade heat exchanger outlet, condenser outlet, and evaporator outlet are saturated.
- Pressure losses in heat pipes and connecting pipes have been neglected.
- The difference between the refrigerated space temperature (T_f) and the evaporation temperature ($T_{\mathrm{Evap,\ CO2}}$) is constant and equal to 5 °C.

The influence of the design parameter on the system's COP and its exergetic efficiency was studied. A prototype of cascade refrigeration system was taken as a reference and the isentropic efficiency of each compressor is taken equal to volumetric efficiency. To solve each case derived from the parametric study, specific software developed by Sieres and Fernandez-Seara [55] has been employed. The thermodynamic properties of CO_2 and ammonia were calculated from REFPROF [56]. To evaluate the exergy losses of each system's components and the exergy loss rate of whole system, a parametric study was applied at the prototype having specified design conditions. The system cooling capacity is 9 kW, the CO_2 evaporator temperature is 5 °C, ammonia condensing temperature is 40 °C and the cascade heat exchanger temperature difference (DT) is 5 °C.

Fig. 14 shows the exergy loss of each system's components and exergy loss rate of the whole system. Results show that the highest exergy loss occurs at the lowest CO_2 condensing temperature and the decreasing trends are present in the all the ammonia system's components. To evaluate the influence of the operating parameters on both system's COP and exergetic efficiency, a statistical procedure has been used to analyze the parametric results. This statistical procedure is the ANOVA Multifactorial [57] and on the basis of it, all evaluated values have the 95% confidence level. Figs. 15 and 16 depict the system's COP and exergetic efficiency values and range of values results from the parametric studies for each of the evaluated

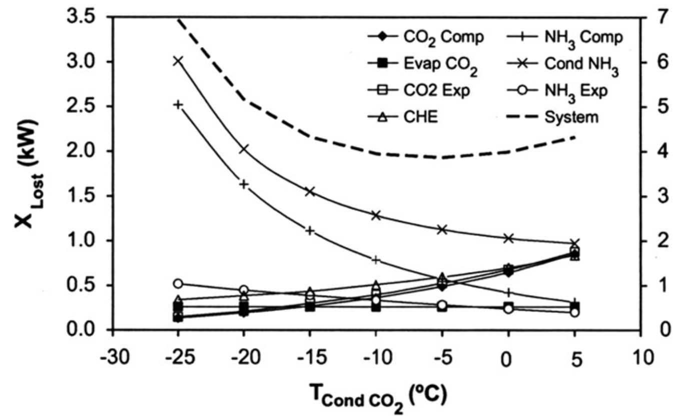

Fig. 14. Exergy lost rates of each system's components(left axe) and of the Whole system (right axe) as a function of T_{Cond},CO_2. Dopazo et al. [54].

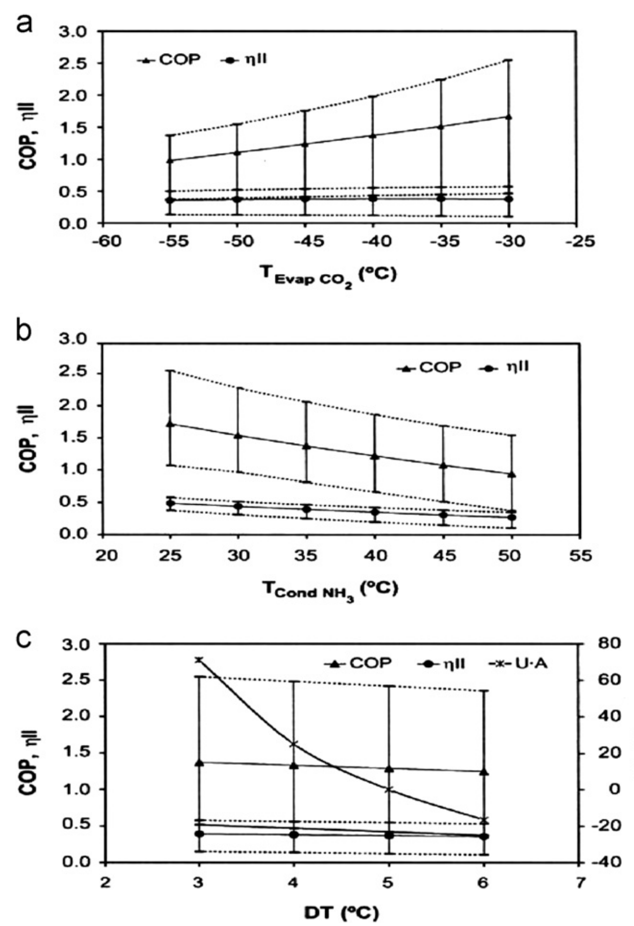

Fig. 15. Avearge values and range of values of the system's COP and exergetic efficiency as a function of (a) $T_{\mathrm{evap,CO2}}$ (B)$T_{\mathrm{Cond\ NH3}}$ and(c) DT. (c) Also includes the relative U.A increments in the cascade heat exchanger as a function of DT. Dopazo et al. [54].

parameters. In Fig. 15, system's COP and exergetic efficiency trends increase with increase in $T_{\mathrm{Evap,\ CO2}}$, $T_{\mathrm{Cond,\ NH3}}$ and DT. The result shows that $T_{\mathrm{Evap,\ CO2}}$ has greater effect than $T_{\mathrm{Cond,NH3}}$. The exergetic influence depicted also shows that there is an initial increasing up to −40 °C and further efficiency shows decreasing trends with $T_{\mathrm{Cond,NH3}}$ and DT.

Fig. 16 shows the COP variations and exergetic efficiency behavior versus $T_{\mathrm{Cond,\ CO2}}$, and shows the increasing trends initially up to −5 °C and afterwards shows a decrease. From the results obtained, a clear viewpoint of the method to obtain maximum COP

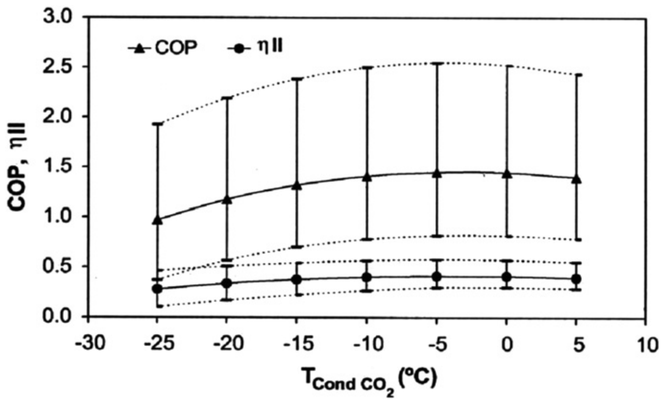

Fig. 16. Average value and range of the values of the system's COP and exergetic efficiency as a function of T_{Cond}, CO_2. Dopazo et al. [54].

value is proven. The CO_2 temperature has to be as high as possible in order to obtain highest COP and it depends on the environmental temperature which has to be cooled, and can be achieved by reducing the temperature difference between the evaporator and the temperature of the area that has to be cooled. On the other hand ammonia condenser temperature has to be as low as possible to maximize the COP and with regard to temperature difference in the cascade heat exchanger, it is clear that for a lower DT, COP is the highest.

3. Simulation studies of different types of refrigeration cycles

3.1. Simulation study of the single stage ARS using conventional working fluids and alternatives

Absorption–refrigeration systems (ARSs) are attractive because they can employ natural refrigerants (water, ammonia, methanol etc.) and can be driven by waste heat, geothermal energy, solar energy etc. and they can also help in the reduction of the green house gases such as CO_2 [58–60] by reducing the consumption of fossil fuels. Absorption–refrigeration system uses various types of absorbent–refrigerant pairs. The absorbent is a secondary solution to absorb the primary solution known as refrigerant in the vapor phase. The most widely used working fluid pairs are ammonia–water and lithium bromide–water. The selection of an appropriate working substance in absorption–refrigeration cycle is very important because its properties greatly affect the performance of the cycle. In recent years many great studies have focused on this problem. These studies are mostly carried out on different working pairs, their properties and also on the thermodynamic analysis of the absorption–refrigeration cycles using different working fluids [61–86]. The schematic diagram of single stage ARS is shown in Fig. 17. The theoretical calculations are carried out by Karamangil et al. [87] on the flow ratio (FR) and the COP of the system. These calculations are based on the input values of the operating parameters like temperature (T_G, T_A, T_C, and TE), heat-exchanger effectiveness (ϵ_{SHE}, ϵ_{RHE}, ϵ_{SRHE}), pump efficiency (η_P) and also on the following assumptions given below:

- The systems are simulated under steady conditions.
- The pressure in generator, condenser, evaporator and absorber is equal to the vapor pressure of the working fluid.
- The pressure drop in the pipes and vessels is negligible.
- The solutions leaving the generator and absorber are at same temperature.
- The expansion process in the expansion device is isenthalpic.

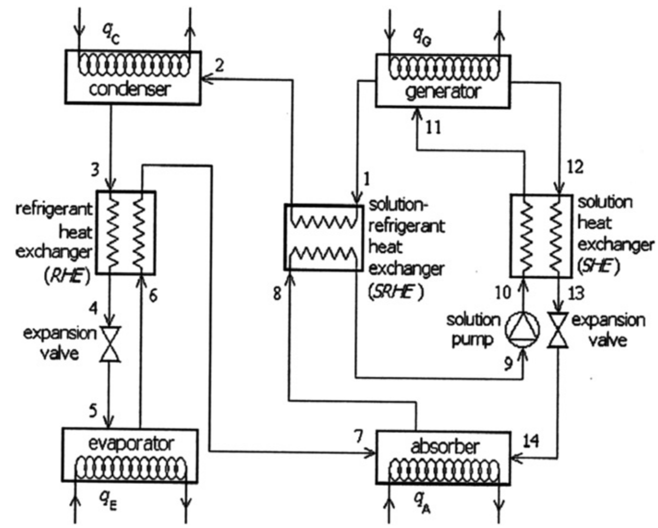

Fig. 17. The schematic illustration of the single stage ARS. Karamangil et al. [87].

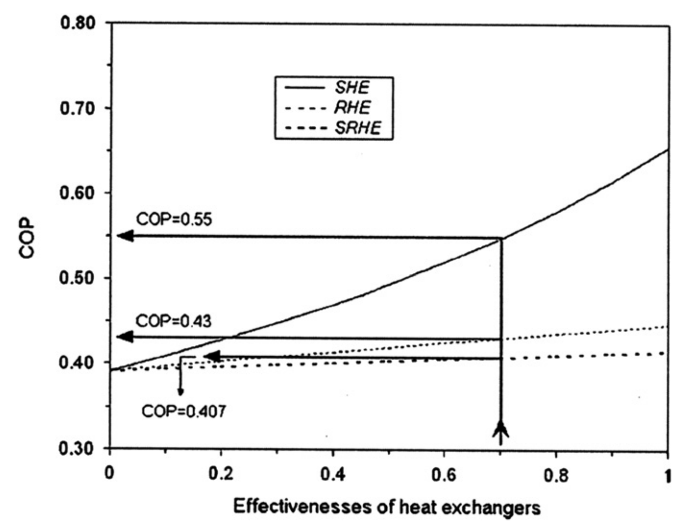

Fig. 18. Variation of COP with effectiveness of heat exchangers. Karamangil et al. [87].

The influence of the effectiveness of the heat exchanger (SHE, SRHE, RHE) on the performance of the system using ammonia–water is shown in Fig. 2. In these simulations T_G=90 °C, T_C=T_A=40 °C, T_E=10 °C and η_P=90% were assumed.

Fig. 18 also shows that the system performance increases gradually with an increase in effectiveness of heat exchanger. But when effectiveness $\epsilon=0$, the COP calculated is only 0.3. With the increase in the heat exchanger effectiveness, the COP increases gradually. The effect of SHE is more pronounced than those of SRHE and RHE and this is due to the increase in the mass flow rates and difference between the inlet temperatures of the fluid. Also with the increase in the effectiveness, the generator temperature decreases and less stress are laid on the heating source thereby improving COP [88,89]. It is also noted that for the theoretical considerations, the effectiveness of heat exchanger is taken as 80% by Sozen [90]. The results of absorption–refrigeration cycle are comparatively presented in Figs. 19 and 20 for LiBr–H_2O, NH_3–H_2O, NH_3–$LiNO_3$ solutions. Flow ratio (FR) is an important parameter in optimizing the system since it directly relates to the cost and size of the generator, absorber, heat exchanger and solution pump [91]. This reveals that with

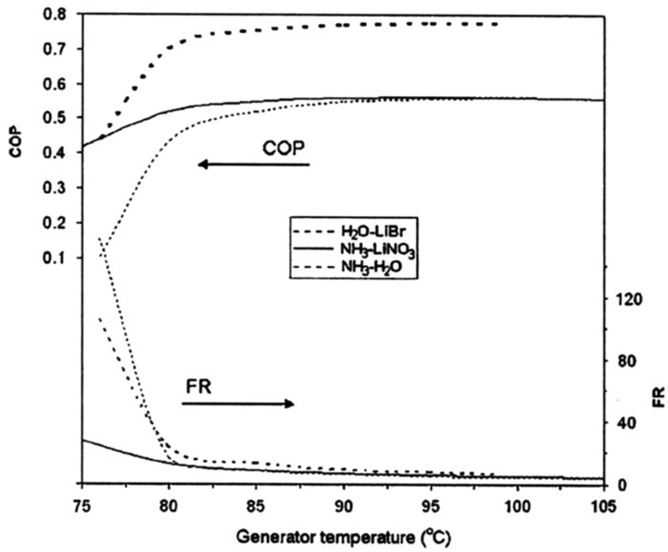

Fig. 19. Variation of the COP AND FR with generator temperature ($T_C = T_A = 40$ °C, $T_E = 5$ C, $\epsilon_{SHE} = \epsilon_{RHE} = 0.70$ and $\eta_P = 0.90$). Karamangil et al. [87].

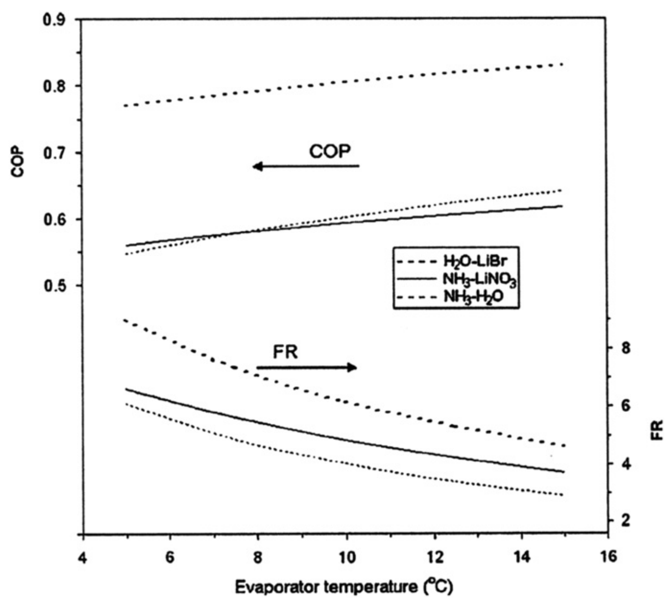

Fig. 20. Variation of the COP and FR with the evaporator temperature($T_G = 90$ °C, $T_C = T_A = 40$ °C, $\epsilon_{SHE} = \epsilon_{RHE} = 0.70$ and $\eta_P = 0.90$). Karamangil et al. [87].

increase in T_G, concentration of the strong solution of H_2O–LiBr increases and concentration of the weak solution of ammonia based mixtures decreases and consequently the COP of the system increases with the increase in T_G.

Variation of flow ratio (FR) and COP values with evaporator temperature is shown in Fig. 20. It is observed that COP values of the cycle increase while FR values decrease with increase in evaporator temperature and these results are quite in agreement with the work done by the others researchers also [69,80].

3.2. Simulation of compressor assisted triple-effect H_2O–LiBr absorption cooling cycles

The COP of the conventional double effect LiBr–H_2O absorption cooling cycle is approximately 1.2. Recently, triple-effect absorption cycle has attracted much interest to replace the existing conventional machine. The expected COP of triple-effect is known to be 30% more than that of the double-effect cycle. A basic type of the triple-effect cycle might be composed by adding one more generator into the existing series or parallel flow double-effect cycle. Many types of the advanced triple-effect cycles have been suggested by [92–102] but the construction of the triple-effect absorption cooling machines using LiBr as the fluid is greatly hindered by the corrosion problem caused by the high generator temperature above 473.15 K. The conventional LiBr–H_2O solution is known to cause the corrosion problems to the metal part of the machine operating at this temperature [103]. More applicable type of the triple-effect cycle could be constructed by coupling two absorption cycles with different solution circuits for low and high temperature [104] but the sorption in the high temperature region for coupling triple/multiple effects could also be realized [101]. One possible way to operate basic LiBr–H_2O triple-effect without corrosion is to lower the generator temperature down to favorable region by using a compression unit i.e. by composing a compressor assisted absorption cycle. These types of hybrid compression–absorption were already considered long time ago [96,97], but extensive discussions have recently suggested using low grade heat for sorption cooling or increasing the temperature lift of the compression heat pumps [102,104,105]. Kim et al. [106] studied a series of computer simulations for four different compressors assisted LiBr–H_2O triple-effect cycles to overcome the corrosion problems. The simulation was focused on lowering the generator temperature using a compressor to investigate the performance characteristics. Simulation of the basic and the compressor assisted systems was carried out numerically by using Engineers Equation Solver (EES) v. 4.999 [39] and embedded thermo-physical data. The main assumptions used in the cycle simulation are as follows:

- All cycles are operated in the steady-state conditions.
- Solution and refrigerant are in equilibrium at the given conditions.
- Refrigerant vapor generated in High Temperature Generator (GH) is totally Condensed and all the evolved heat is transferred to the Medium Temperature Generator (GM) as the energy source.
- Refrigerant vapor generated in the GM is totally condensed and all the evolved heat is transferred to low generator temperature as the energy source.

Several parameters and operating conditions were fixed by specific values as listed in Table 2. Using these parameters and operating conditions, all state point variables including temperature, pressure, concentration, flow rate, vapor quality, enthalpy, and entropy were calculated. The results in Fig. 21 show the decrease of GH temperature against the vapor-compression ratio of each cycle. The GH temperature of all compressor assisted cycles decreases with increasing vapor compression ratio as expected. Type 4 shows the largest decrease of the GH temperature and

Table 2
Operating conditions and fixed parameters adapted for cycle simulation.

Conditions and parameters	Fixed value
Cooling capacity	3.516 kW (1 TR)
Evaporator outlet temperature	278.15 K
Absorber outlet temperature	308.15 K
Condenser outlet temperature	313.15 K
Solution flow rate (absorber outlet)	0.022 kg s^{-1}
Temperature differences for heat transfers from condensed refrigerants to GM and GL	4 K
Absorber loss	2 K
Efficiency of all heat exchangers	0.8
Isentropic compressor efficiency	0.8

type 3 shows the smallest at the same compression ratio. This is because generator temperature decreases with the introduction of the compression unit and hence results in the decrease in the heat input to the system too. Fig. 22 represents the work for the compressor operation along with the corresponding heat input, as the function of the GH temperature decrease. Thus Type 1 requires the largest compressor work and type 4 the smallest for the same degree of GH temperature decrease (heat source) and hence Type 4 needs the lower input and vice-versa.

With the increase in the vapor-compression ratio the generator temperature could be extensively lowered. The mechanical energy input required for the compressor operation reduced the heat energy input required for constant cooling power and can be seen in Table 3.

3.3. Modeling and performance study of a continuous adsorption refrigeration system driven by a parabolic trough collector

Adsorption–refrigeration systems are environmentally friendly alternatives to the other existing systems, since they can use refrigerants that do not contribute to ozone layer depletion and global warming. They also operate at lower costs in comparison with the absorption systems, as they do not need solution pump. The refrigeration as solar energy application is particularly attractive because of non-dependence on conventional power. Despite their advantages, the adsorption–refrigeration systems have some drawbacks, such as, low COP, low SCP, high weight and high cost. In order to recover such shortcomings various researchers have worked on the improvements of the adsorption systems. During recent decades, several solar adsorption–refrigeration units were tested with different combinations of adsorbents and adsorbates. The most common pairs studied are carbon/ammonia, activated carbon/methanol, zeolite/water, silica gel/water. These investigations include research on ice-making and shows adsorption and cooling only during nights [107–113]. Most of the studies conducted on the solar adsorption cooling systems have been achieved with either flat-plate or evacuated tube collectors, whereas little attention has been devoted to concentrating collectors, in particular the PTCs. A schematic description of the proposed two-bed continuous adsorption–refrigeration system is shown in Fig. 23. The mathematical analysis is done by Fadar et al. [114]. The analysis made is based on few model assumptions:

- Pressure inside the absorbent bed is uniform.
- The absorbent bed is considered as continuous and the heat transfer in the medium is characterized by an equivalent thermal conductivity.

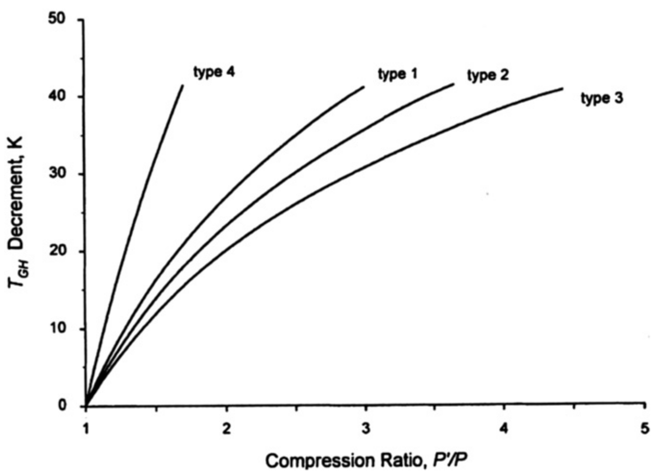

Fig. 21. Decrease of GH temperatures as function of the compression ratio. Kim et al. [53].

Table 3
Performance characteristics of basic and compressor-assisted [a] cycles [b].

Cycles	T_G^c (K)	Q_E (kw)	Q_G^c(kw)	W_{Comp}(kw)	COP [therm]	COP[mech]
Single-effect	359.58	3.516	4.70	–	0.75	–
Double-effect	413.45	3.516	2.85	–	1.24	–
Triple-effect	475.95	3.516	2.28	–	1.54	–
Type 1	435.95	3.516	2.04	0.194	1.72	18.1
Type 2	435.95	3.516	2.06	0.171	1.71	20.6
Type 3	435.95	3.516	2.07	0.173	1.70	20.4
Type 4	435.95	3.516	2.03	0.125	1.74	26.7

[a] At 40 K's GH temperature decreases.
[b] All cycles have same intenal temperature condition of evaporator, condenser, absorber.
[c] Temperature and heat input in the highest generator temperature.

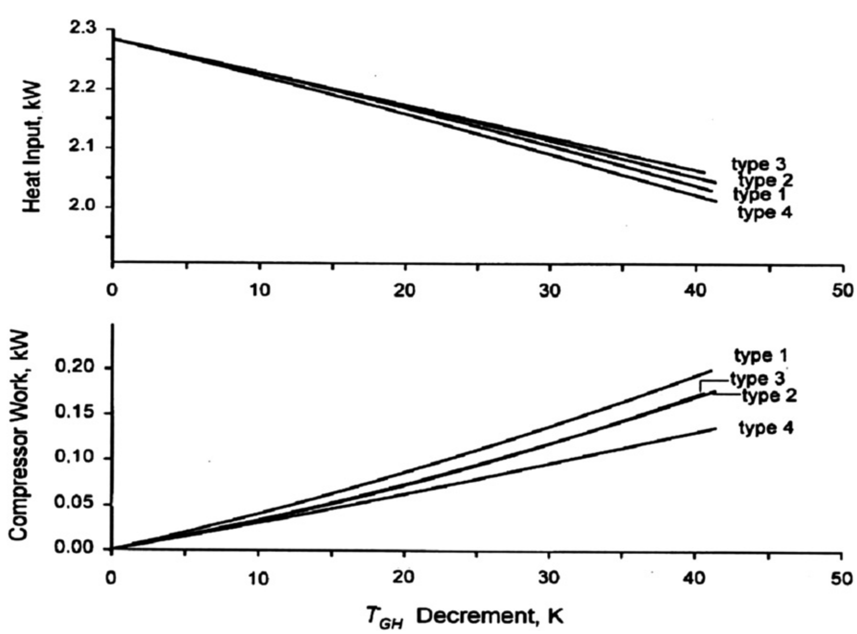

Fig. 22. Heat and compressor work inputs of compressor-assisted cycles. Kim et al. [53].

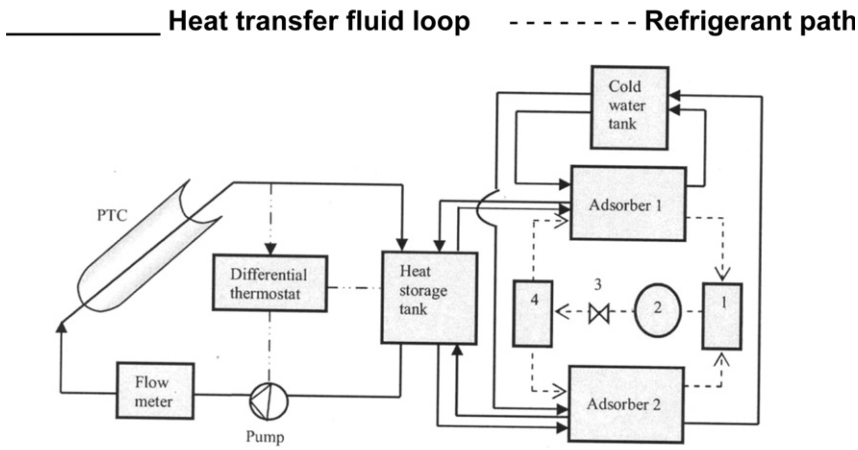

Fig. 23. Schematic diagram of the solar powered continuous adsorption–refrigeration system (1) condenser, (2) ammonia tank, (3) expansion valve evaporator. El Fadar et al. [114].

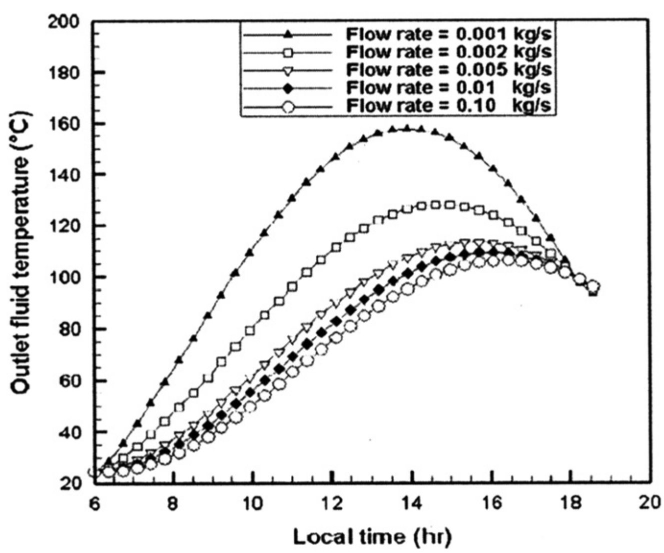

Fig. 24. Effect of mass flow rate on outlet fluid temperature (V_{St}=0.015 m^3). El Fadar et al. [114].

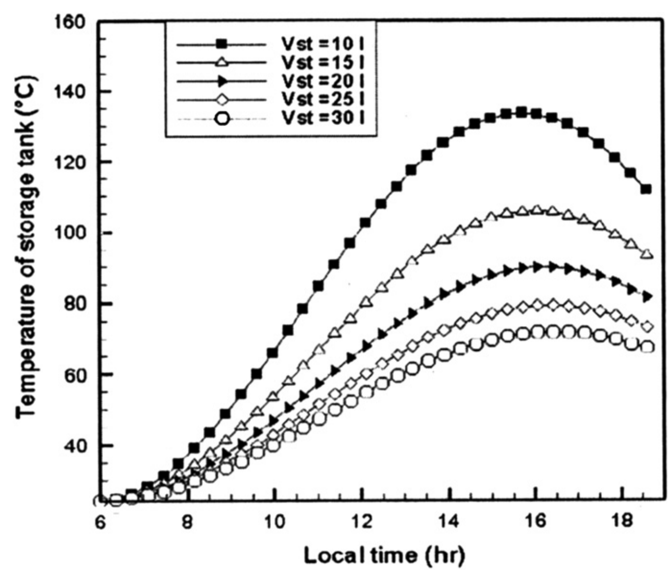

Fig. 25. Daily variation of mean water temperature in storage tank with different tank volumes. El Fadar et al. [114].

- All phases are continuously in thermal, mechanical, chemical local equilibrium.
- The condenser and the evaporator are ideal i.e. T_{Evap} and T_{Cond} are constant during the isobaric process and adsorption/desorption process is an isobaric process.
- The two absorbers have identical thermo-physical, structural, and geometrical characteristics.

A one dimensional model based on combined heat, mass transfer and thermodynamics is framed considering the assumptions stated above. The assessment parameters of the performance of adsorption–refrigeration system considered in this study are the refrigeration COP (COP $_{cycle}$), the solar coefficient of performance (COPs) and the specific cooling power (SCP). The discretized equations are solved using the Tri-Diagonal Matrix Algorithm (TDMA) and the non-linearity of the equations is solved by iterative techniques. A computer program written in FORTRAN has been developed in order to simulate the behavior of the adsorption cooling system. The effect of the mass flow rates on outlet fluid temperature is depicted in Fig. 24. It can be seen that with the increase in mass flow rate, the outlet fluid temperature reduces and is more pronounced for small values (0.005 Kg/s). Fig. 25 represents the daily variation of the mean water temperature in the heat storage tank (T_{St}) with different storage tank volumes (V_{St}).

Under the same conditions of mass flow rate (0.01 Kg/s), Fig. 26 shows the effect of heat storage tank size on both maximum temperatures that could be reached in the tank and heat stored in this tank (Q_{St}). As expected greater the heat storage tank size, lower is the maximum temperature of the storage tank and more the heat stored in the tank. The optimal size of the heat storage tank would range from 0.02 to 0.05 m^3. The variations of the COP_{cycle}, SCP_{cycle} with absorbent bed thickness and heat source temperature is shown in Fig. 27. It is quite clear that with the increase in the bed thickness the COP_{cycle} remains almost the same. On the contrary, with the increase in the bed thickness the resistance for the heat flow increases and hence the SCP decreases. The result in Fig. 28 depicts the variation of the daily cooling production and COP with adsorbent bed thickness. It is observed that under daily solar radiation of 14 MJ in 0.08 m^2 collector area, the COP and cooling power decreases from 0.18 to 0.082. This is because with higher values of bed thickness the cycle time becomes relatively longer. The thickness beyond 40 mm is not recommended.

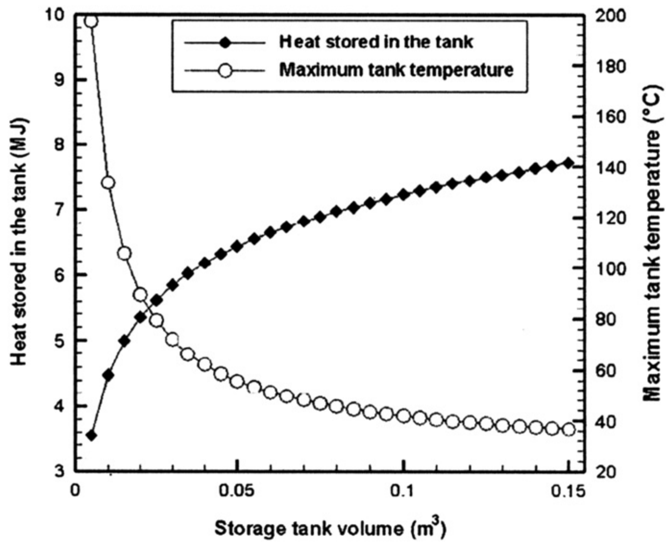

Fig. 26. Variation of maximum tank temperature and heat stored in the tank with storage tank volume. El Fadar et al. [114].

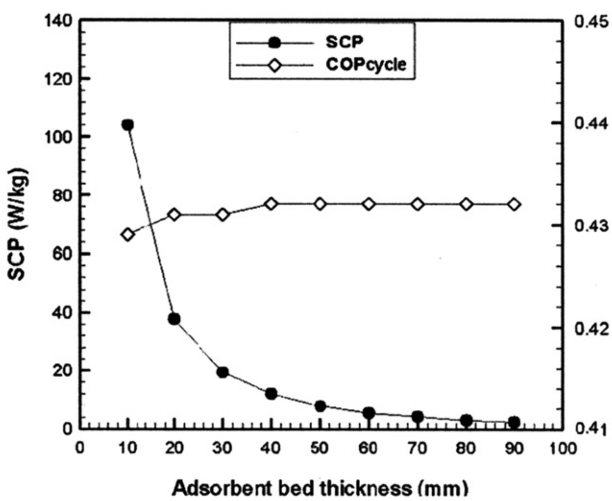

Fig. 27. Influence of the adsorbent bed thickness on SCP and COP_{Cycle} (T_{Heat} = 100 °C). El Fadar et al. [114].

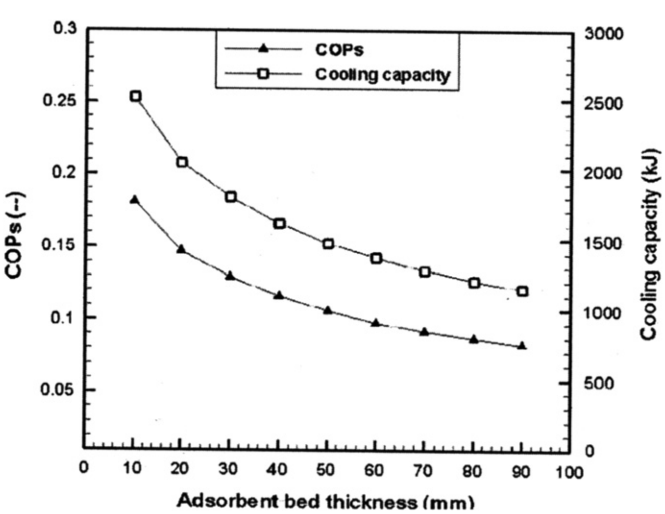

Fig. 28. Variation of daily cooling capacity and COP's with absorbent bed Thickness (T_{Heat} = 100 °C). El Fadar et al. [114].

3.4. Mathematical simulation of solar ejector-compression refrigeration system

An ejector refrigeration cycle is shown in Fig. 29. The cycle consists of two sub cycles, the generator part is power cycle and the evaporator part is a refrigeration cycle. The convergent–divergent type of nozzle in the ejector is used for expansion of refrigerant. The COP and thermodynamic behavior of cycle therefore depends on the ejector characteristics. Fig. 30 represents the influence of the generator temperature (T_G) on the COP for a given ejector geometry [115].

The production of ice requires evaporator temperature of at least −10 °C and for this it needs the compression ratio greater than 4 which is out of the range for the ejector compressor. This problem was removed by Sokolov and Hershagal [116,117], by introducing a hybrid cycle for ejector compressor using a booster. There are many practical results obtained for these systems using solar energy as an energy source [118,119].

Dorantes et al. [120] carried out mathematical simulation and chose R142b as the working fluid for ejector compression refrigeration system. The whole system operates depending on values of storage and ambient temperatures (T_a, T_s) keeping the evaporator temperature constant. The fluid has adequate thermodynamic properties to the imposed system operating conditions. The refrigeration cycle operates at condenser temperature (T_C) of 30 °C, generator temperature (T_G) of 105 °C, an evaporator (T_E) of −10 °C, with a required generator heat (Q_G) of 5.6 KW and an obtained evaporator heat (Q_E) of 2 kW; the corresponding COP obtained was 34%. This generation heat which is the function of the condenser temperature is calculated from method developed by Dornates [121]. With these conditions, the ejector geometry

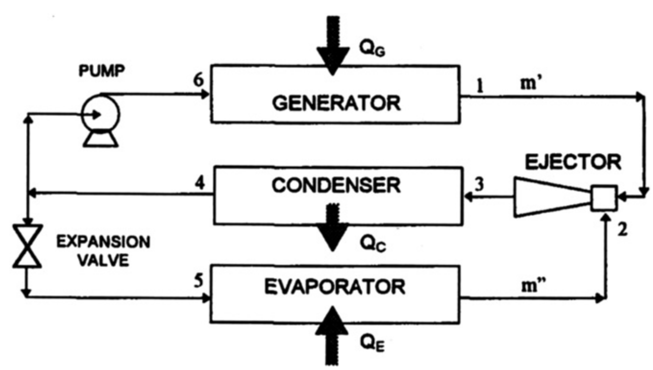

Fig. 29. A Block diagram of the ejector-compression refrigeration cycle. Dorantes et al. [120].

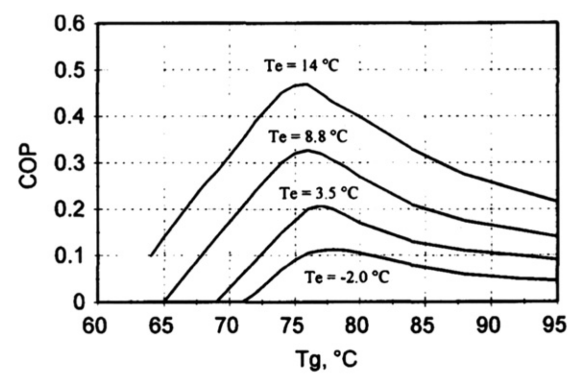

Fig. 30. Influence of generator temperature (T_G) on the COP for a given ejector geometry with fluid R-11. Dorantes et al. [120].

was fixed and the curves for Q_G, Q_E and COP as function of T_C and T_G are obtained.

Fig. 31 shows the daily history of the system storage temperature for two days of the year i.e. 17 and 162, which corresponds to the months of January and June. Also the Q_U and Q_{Load}, heats are plotted for the same days. As expected, in both cases temperature and heat are higher in June than in January.

Fig. 32 shows the monthly average COP, collector and system efficiencies and also the monthly average ice production. It is evident that the lower efficiencies were obtained in the summer because of increase in the ambient temperature (T_a). In contrast the ice production increases in the winter and reaches its maximum value. Despite the climatic variations, the annual average value for COP was 21%. The annual average collector efficiency and system efficiency were 52% and 11% respectively.

3.5. Simulation of trans-critical CO_2 heat pump cycle for simultaneous cooling and heating applications

A CO_2 based vapor compression refrigeration system came into existence some 100 years back. However due to problems arising from low critical temperature and high operating pressure, it was slowly replaced by other refrigerants. Extensive applications of CO_2 heat pump were reported by many authors [122,123]. They have extensive applications in two sectors—(a) mobile air conditioning and, (b) heat pumps for simultaneous heating and cooling. Although several simulation studies of conventional vapor compression systems have been reported [124–126]. A carbon dioxide based heating and cooling system was analyzed by Sarkar et al. [127] showing the main components in Fig. 33. The entire system has been modeled based on energy balance on individual components. The following assumptions have been made in the analysis:

- Heat transfer with the ambient is negligible.
- Only single phase heat transfer occurs for water.
- Compression process is adiabatic but not isentropic.
- Pressure drops on water side and in connecting pipes are negligible.

A computer code has been developed to simulate the trans-critical CO_2 system, using new equations of state for CO_2 [105] and a transport property correlations available in the literature [106]. The performance of the system being studied for simultaneous heating and cooling applications is evaluated on the basis of system COPs which have been estimated for various water inlet temperatures for heat exchangers, compressor speed and heat exchanger dimensions. Figs. 34–37 show the effect of varying area ratio and operating conditions for the same evaporator length and gas cooler length. The variations of cooling load, compressor work and system COP with area ratio of water inlet temperature of 30 °C and rated speed of 2900 rpm are shown in Fig. 34 and corresponding optimum pressure and refrigerant mass flow rate variations are shown in Fig. 35. Results show that with increase in area ratio and at optimum discharge pressure, cooling output and compressor work reduce due to decrease in refrigerant flow rate. The decrease in mass flow rate is due to decrease in

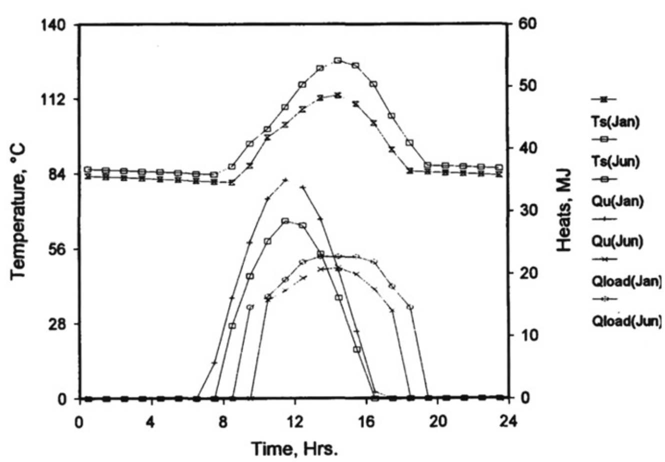

Fig. 31. Daily history of system storage temperature for two days of the year i.e.17&162.The Q_U and $Q_{Load}=Q_G(T_C,T_{St})$ heats are plotted for the same days. Dorantes et al. [120].

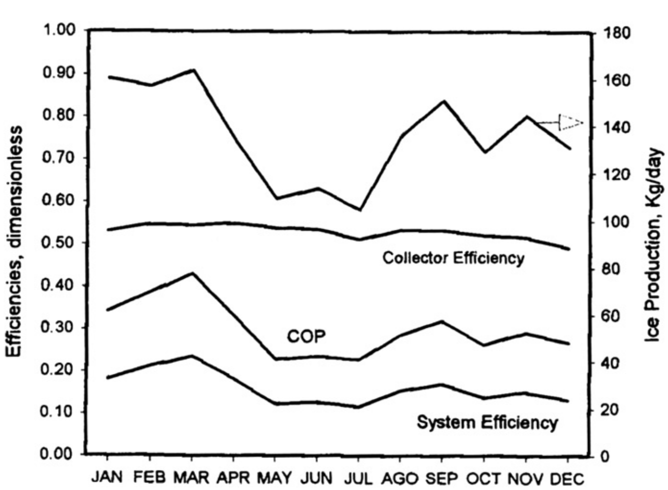

Fig. 32. Monthly average COP,collector and system efficiencies and the monthly average ice production for the 12 months. Dorantes et al. [120].

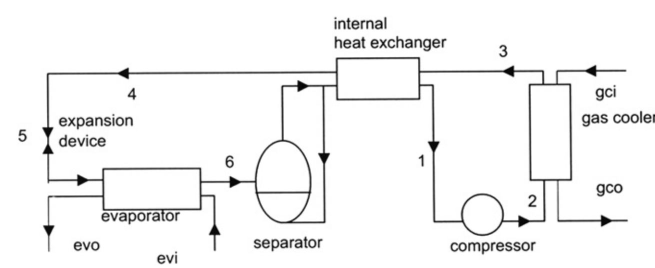

Fig. 33. Schematic layout of a transcritical carbon dioxide system. Sarkar et al. [127].

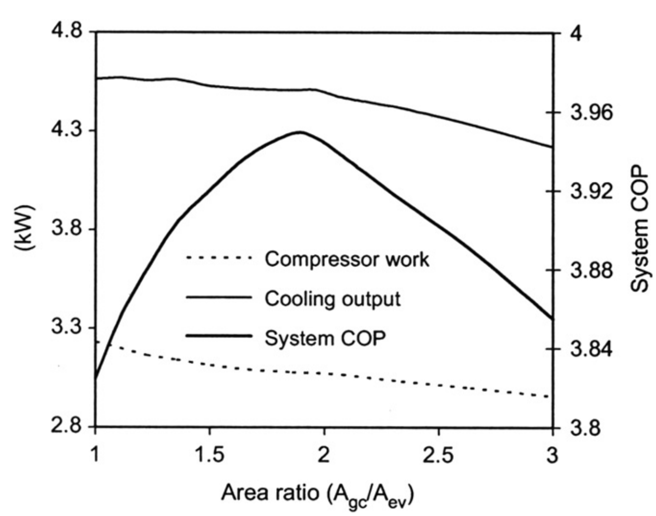

Fig. 34. Variation of performance with area ratio. Sarkar et al. [127].

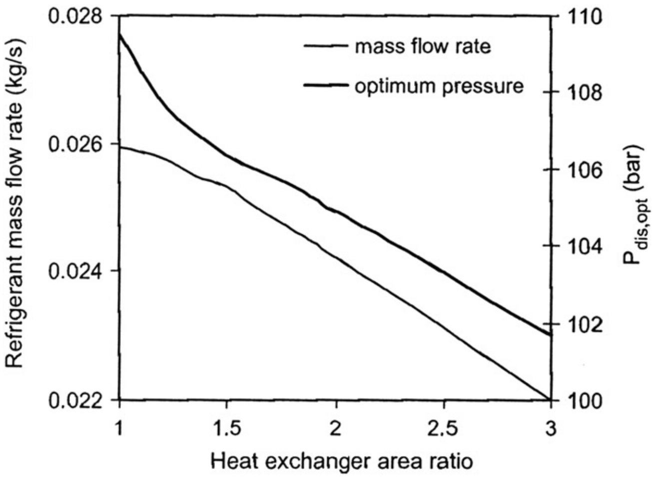

Fig. 35. Variation of optimum discharge pressure and mass flow rate with area ratio. Sarkar et al. [127].

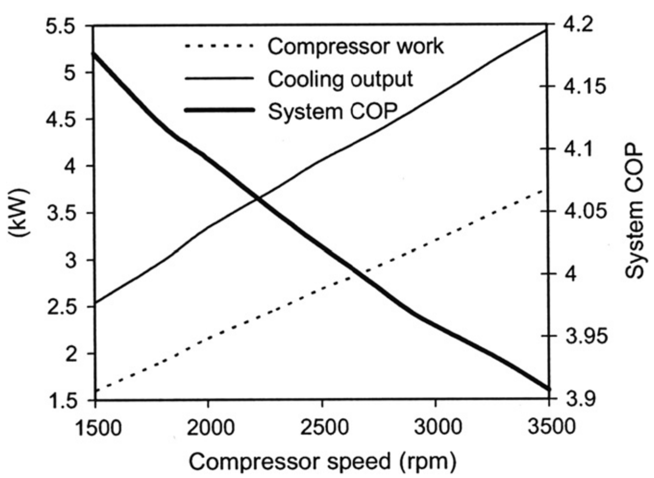

Fig. 36. Variation of performance with compressor speed. Sarkar et al. [127].

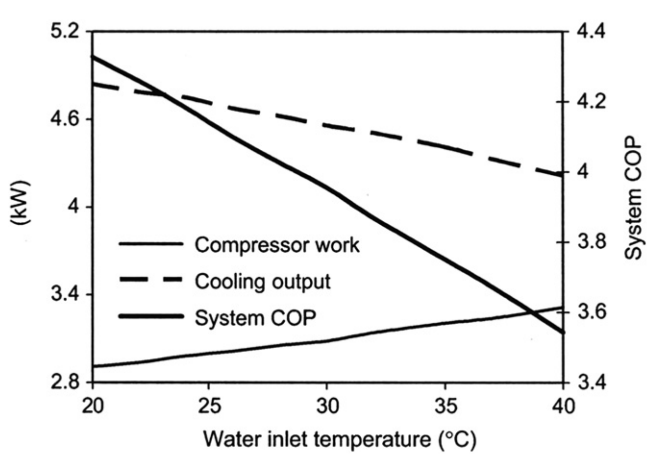

Fig. 37. Variation of performance with water inlet temperature. Sarkar et al. [127].

optimum discharge pressure as shown in Fig. 36. The effect of the compressor speed on the system performance at heat exchanger area ratio of 1.85 and water inlet temperature of 30 °C is presented in Fig. 37. It is observed that the system COP at optimum pressure decreases with the compressor work and the cooling output which however increases with the compressor speed.

4. Discussion of results

The results of the modified aqua-ammonia absorption system for the variation of COP with the generator temperature for two values of evaporator temperature i.e. 0 °C and −10 °C revealed that the variation of COP with the generator temperature is higher at higher condenser temperature. The results of the thermodynamic study of HFC134a-DMA based double-effect cascaded refrigeration system reveals that Double-effect HFC134a-DMA absorption systems yield COP comparable to single-stage HCFC22-DMA systems, but at slightly higher heat source temperatures and to obtain refrigeration at sub-zero temperatures at low heat source temperatures, one may have to resort to cascade systems. The analysis of a simple LiBr–H_2O absorption cooling system shows that the system operating at the generator temperature of 90 °C becomes more efficient in providing the evaporator temperature of 4.5 °C and provides good value of COP indicating an option of solar energy as an appropriate source of energy for the absorption systems. The energy and exergy analyses of an actual vapor compression cycle operating with R502, R404A and R507A as working fluids indicate that COP and exergetic efficiency for R507A are better than those for R404A at condenser temperatures between 40 °C and 55 °C, but both refrigerants show 4–17% lower value of COP and exergetic efficiency in comparison to R502 for the condenser temperatures between 40 °C and 55 °C. Hence, the increasing order for the refrigerant for improved COP as well exergetic efficiency is as R404A, R507A and R502. The study also indicates that the worst component from the viewpoint of irreversibility or exergy destruction is condenser followed by compressor, throttle valve and evaporator. The increase in dead state temperature has a positive effect on exergetic efficiency and EDR, i.e. EDR reduces and exergetic efficiency goes up with increase in dead state temperature. The comparative thermodynamic analysis of single stage and two stage thermal compression with mechanical compression indicates that COP of a two-stage system almost remains constantly fairly uniform over the entire range of evaporating temperatures investigated, single-stage system is better than the two-stage system for evaporator temperature greater than 5 °C and COP of the hybrid cycle is the highest because of less energy needed for second stage which is mechanical compression. The results also indicate that the packing effectiveness shows a qualitative improvement in the performance. By comparison of the uptake efficiencies for the first as well as the second compressors in a two stage system, the uptake efficiencies of the hybrid system are approximately equal to the variation of volumetric efficiency. Thus, hybrid compression can be a potential avenue of energy conservation in refrigeration systems. The theoretical analysis of a CO_2/NH_3 cascade refrigeration system for cooling application at low temperatures indicates that system's COP and exergetic efficiency trends increase with increase in $T_{\mathrm{Evap,\ CO2}}$, $T_{\mathrm{Cond,\ NH3}}$ and DT, but $T_{\mathrm{Evap,\ CO2}}$ has greater effect than $T_{Cond,\mathrm{NH3}}$ because $T_{\mathrm{Cond,NH3}}$ does not show increasing trend beyond −40 °C. The results indicate that the CO_2 temperature has to be as high as possible in order to obtain the highest COP and it depends on the environmental temperature which has to be cooled, and can be achieved by reducing the temperature difference between the evaporator and the temperature of the area that has to be cooled. On the other hand ammonia condenser temperature has to be as low as possible to maximize the COP and with regard to temperature difference in the cascade heat exchanger, it is clear that for a lower DT, COP is the highest. The simulation study of the single stage absorption–refrigeration system using conventional working fluids and alternative reveals that system performance increases gradually with an increase in

effectiveness of heat exchanger. The effect of SHE is more pronounced than those of SRHE and RHE and this is due to the increase in the mass flow rates and difference between the inlet temperatures of the fluid. Also with the increase in the effectiveness, the generator temperature decreases and less stress is laid on the heating source thereby improving COP. It is observed that COP values of the cycle increase while FR values decrease with increase in evaporator temperature. The simulation of compressor assisted triple-effect LiBr–H_20 absorption cooling cycles reveals that GH temperature of all compressor assisted cycles decreases with increasing vapor compression ratio as expected. This is because generator temperature decreases with the introduction of the compression unit and hence results in the decrease in the heat input to the system too. The performance study of a continuous adsorption–refrigeration system driven by a parabolic trough collector reveals that with the increase in mass flow rate, the outlet fluid temperature reduces and is more pronounced for small values. The effect of heat storage tank size on both maximum temperatures that could be reached in the tank and heat stored in this tank (Q_{St}) is analyzed and it clearly indicates that greater the heat storage tank size, lower is the maximum temperature of the storage tank and more the heat stored in the tank. The COP_{cycle}, SCP_{cycle} with absorbent bed thickness and heat source temperature indicates that with increase in the bed thickness COP_{cycle} remains almost the same. On the contrary, with increase in the bed thickness the resistance for the heat flow increases and hence the SCP decreases. The simulation of a solar-ejector compression refrigeration system indicates that system storage temperature as well as heat is higher in the months having more ambient temperature. The results also indicate that monthly average COP, collector and system efficiencies become lower because of increase in the ambient temperature (T_a). In contrast, the ice production increases in the winter and reaches its maximum value. The simulation study of a trans-critical CO_2 heat pump cycle for Cooling and Heating Applications indicates that with an increase in area ratio and at optimum discharge pressure, cooling output and compressor work reduces due to decrease in refrigerant flow rate. The decrease in mass flow rate is due to decrease in optimum discharge pressure. It is observed that the system COP at optimum pressure decreases with the compressor work and the cooling output which however increases with the compressor speed.

5. Conclusion

The present study investigates the utility of the simulation techniques available to analyze the refrigeration cycles. The detailed study was carried out to find out the influence of different operating parameters on the performance of refrigeration cycles and the following conclusions are drawn:

- The simulation results showed that the COP values of the cycle increase by increasing generator and evaporator temperature, but decrease by increasing condenser and absorber temperature as expected.
- The generator temperature has a great effect on the COP and the use of the heat exchanger has a great effect on the improvement of the COP. The use of the SHE improved efficiency up to 60%.
- However, the systems operated with alternative sources of energy are also considered and it reveals that the non-conventional sources of energy are feasible enough to work as a source of energy to the generator to carry out the cooling performance and is a promising strategy to reduce the harmful gases emissions.
- The worst component from the viewpoint of irreversibility or exergy destruction is condenser followed by compressor, throttle valve and evaporator. The most efficient component is liquid vapor heat exchanger.
- The increase in dead state temperature has a positive effect on exergetic efficiency and EDR, i.e. EDR reduces and exergetic efficiency goes up with the increase in dead state temperature.
- The variation of the absorbent bed thickness affects the specific cooling power more than the cycle COP. The heat source temperature results in the increase in the cycle COP, Solar COP and specific cooling power.

The modification in the components like compressors also helps in improving efficiency of the system. These modifications can be very helpful in reducing the corrosion problems and also some other problems, relating to the interactions with lubricating oil, inefficient heat transfer including super heated vapors, possible salt entertainment in the high pressure vapor stream, too large vapor specific volumes on the suction side of the compressor and so on, should be more closely investigated and minimized. It is hoped that many operations on absorption–refrigeration cycles such as selection of an appropriate working solution pair with respect to operating conditions, the accurate and fast calculations of performance parameters, and the determination of the suitable operating conditions could be done by the developed user friendly software packages. This study can be useful source since it includes a review of the refrigeration cycles and the detailed thermodynamic analysis of the systems and comprehensive information related to the conventional working fluids and alternatives.

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